EXPERIMENTAL MODAL ANALYSIS OF VEHICLE EXHAUST SYSTEM TO DETERMINE HANGER LOCATION USING ROOT MEAN SQUARE VALUE

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ABSTRACT

Vibration of exhaust system due to the engine operation and the condition of the road surface which is transferable to the body through the hanger affects the noise, vibration and harshness performance of the vehicle, and the life time of the hangers. In order to solve this problem of the automobile exhaust system, experimental modal analysis was carried out by utilizing a sample exhaust system of passenger car as a research object within the frequency of interest. LMS Test.Lab was used to perform modal analysis. This study estimated the dynamic characteristics of the exhaust system such as natural frequencies and mode shapes. Using frequency response function at every measuring point, root mean square value (RMS) of vibrating energy was calculated to select hanger location of exhaust system where the RMS value is comparatively small. Experimental modal analysis is an efficient tool to study and estimate the dynamic properties of the exhaust system, and calculating RMS value is also an easy method to find hanger location of the exhaust system utilizing frequency response function.

Keywords— *Hanger Location, Experimental Modal Analysis, Root Mean Square (RMS) Value, Exhausts System, Frequency Response Function*

1. INTRODUCTION

Exhaust system is suspended from the car body by hangers. It is mainly composed of exhaust manifold, exhaust pipe, flexible joint and muffler. The objectives of automobile exhaust system is used to prevent exhaust gas from entering the passenger compartment, to carry and reduce releasing toxic gases into the environment and to silence noise that is occurred by the high pressure exhaust gas. It has an interface between the engine, vehicle body and environment. Due to the excitation of engine operation and the road surface condition, the vibration of exhaust system occurs and transferred to the car body through the hangers. This behavior affects the noise, vibration and harshness performance of the vehicle, and the life time of the hangers with fatigue. In order to solve this problem, the sensitivity locations of hangers are important, and depend on the dynamic characteristics of the exhaust system and the excitation frequency range, especially engine idle frequency [1]. Previously, many works with respect to the modal analysis of vehicle exhaust system have been studied. Recently, Wang Wenzhu et al. arranged hanger location layout of automotive exhaust system using ADDOFD method by numerical and experimental modal analysis. Their constrain modal analysis also depicts the natural frequencies of exhaust system is out of the engine's excitation frequency range [2]. He Hai et al. performed modal testing and estimated the dynamic properties of the exhaust system. Based on the results, they suggested that the overall stiffness should be enhanced and the research on the location of hangs is their further study [3]. Heiner Storck et al. studied experimental modal analysis of exhaust structures providing three boundary conditions [4]. Chetan D. Gaonkar determined location of mounting clamps identifying nodal and anti-nodal points in mode shapes of exhaust system by modal analysis [5]. The finite element method based simulation technique to determine hanger position of exhaust system using average driving DOF displacement (ADDOFD) method was presented by Noorazizi et al [6, 7]. John George et al. carried out modal transient dynamic analysis with vibration data from road load data acquisition (RLDA) which influence the dynamic behavior of exhaust system. Their study shows that attention should be given to exhaust system frequencies below 15 Hz although design engineer focus on the engine idle frequency range (typically 20-40 Hz) to avoid any potential NVH problems [8].

In this study, experimental Modal Analysis was conducted by utilizing LMS Test.Lab testing software within the frequency range (0Hz – 50Hz). The natural frequencies and mode shapes of the exhaust system were generated. Although ADDOFD method was used in the most of the researches, in this paper, root mean square (RMS) value of vibration energy was calculated based on frequency response function at every node. Calculating RMS value can easily and quickly identify where the vibrating energy is small to select hanger location. The main objective of selecting hanger location is to reduce the vibration of the exhaust system for long life use of exhaust hangers and to avoid the noise, vibration and harshness (NVH) issue of the automobile due to the exhaust system.

In this paper, hanger locations where RMS value is relatively small were selected by utilizing a sample exhaust system of passenger car as a research object. This study can contribute to the design development and verifying the performance of an automobile exhaust system and can be broadened for the other exhaust system designs.

2. EXPERIMENTAL MODAL ANALYSIS

Modal analysis is vital to compute the dynamic characteristics of the exhaust system. Experimental modal analysis is a technique to obtain modal parameters of the structure such as natural frequency, damping and mode shape identifying the measured input force and response by exciting a force to the structure.

2.1 Testing Object

The testing object that is used in this study is shown in Figure-1. The components of this exhaust system are flexible bellow, catalyst converter and exhaust pipe.

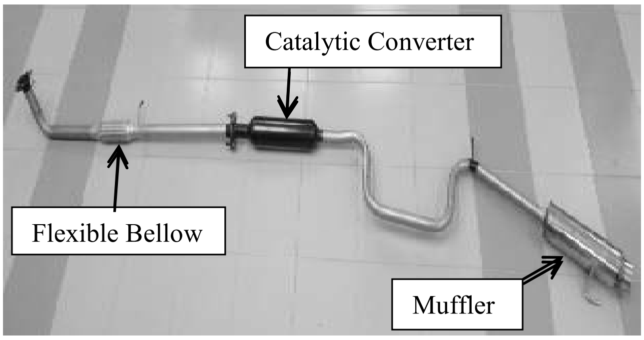

Figure-1. Sample of Testing Object

2.2 Experimental Setup

The experimental setup was performed by the following aspects.

(1) The interested boundary condition of the exhaust system was free-free condition. Very small diameter of fishing cord was used to simulate free-free model of the exhaust system. This cord has very small stiffness and much lower frequency than the lowest frequency of the exhaust system. Five hanging points were chose to be balance including two points that are at front edge and rear edge. The effect of these suspension cords can be negligible.

(2) The excitation point was selected based on the following considerations. The shaker should be placed at the point where the excitation signal can be transmitted the whole exhaust system easily. This point should have high stiffness to make sure the shaker support is resonance free. Since the exhaust system is too long, a point at the middle of the exhaust system was chose as a reference point to get enough signal and response level from all measurement points. A three millimeter diameter and seven centimeter long of stinger rod that should be rigid in the axial direction and flexible in the lateral direction was used in this experiment to decouple the shaker from the testing object and transmit force to the testing object. In order to measure and transmit force to the structure, a force transducer was used at the top of the stinger.

(3) The measuring point should be taken along the profile of the exhaust system to specify modal properties. A total

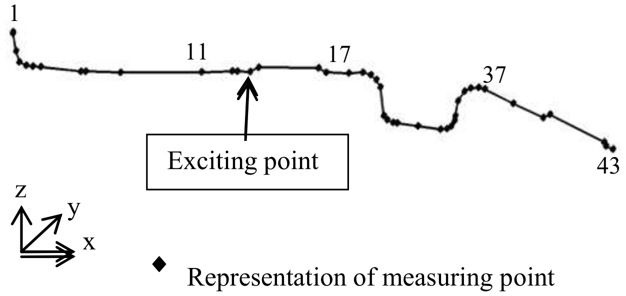

Figure-2. Wireframe of Exciting Point and Measuring Points

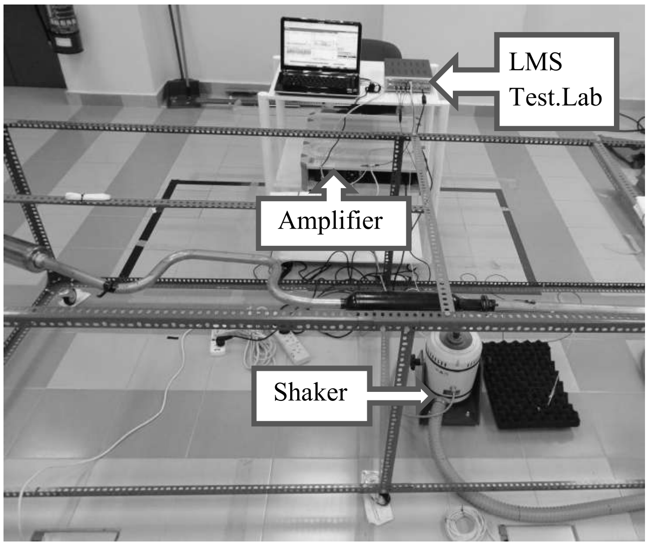

Figure-3. Photo of Testing Setup

of 43 measuring points was selected to simulate the exhaust system profile. The wireframe of exciting point and measuring points are shown in Figure-2. Tri-axial accelerometer that is used in this testing can measure vibration acceleration in X, Y and Z directions of every measuring point but we must ensure that the directions of the accelerometer were the same as the previous measuring point when roving the accelerometer from measuring point to point.

(4) Spectral testing program that is suitable for shaker testing in LMS Test.Lab software was used to obtain more precise vibration properties. Although there are different types of signal such as sine, step sine, random and burst random with shaker excitation, random signal was selected as an excitation signal to approximate nonlinear properties of the exhaust system into linear properties, and another reason is that the excitation signal from the engine vibration and vibration due to the road surface condition which is transmitting through the wheel, suspension system, chassis and hangers to the exhaust system cannot be sinusoidal. In addition, 10 times linear amplitude average method was used to reduce random error and also Hanning window was utilized to avoid the leakage of random signal.

(5) Since the testing object is from passenger car, the frequency range of interest is based on the engine specification of the car which has 20Hz-30Hz of idle frequency. In order to cover engine idle frequency and some vibration frequencies due to the road surface condition, frequency range (0Hz-50Hz) was selected to conduct modal analysis of the exhaust system.

The photo of experimental setup is shown in Figure -3.

2.3 Data Processing

Some initial setup such as channel setup, scope and test setup were completed in the spectral testing software of LMS Test.Lab acquisition system. Then, the measured data were processed in the modal data selection and time MDOF worksheets. The modal parameters were identified from the sum frequency response function of all measuring points. The identified natural frequencies and damping are shown in table 1, and the first five order mode shapes are depicted in Figure-4. Among five numbers of modes, modal orders one to two are bending modes and the rest are complex mode shape. These show that the higher natural frequency, the more complex mode shape occurs and then the displacement distribution of each mode is different meaning that vibration energy at each node is different.

Table-1. Natural Frequencies of Exhaust System

Modal order	Natural frequency, Hz	Damping, %
1	6.975	4.76
2	15.26	4.51
3	20.802	3.81
4	27.533	3.71
5	49.139	0.98

3. DETERMINATION OF HANGER LOCATIONS

3.1 Root Mean Square (RMS) Theory

A square root of the mean value of square quantity *f(t)* at a proper average time *T* is termed the root mean square value for a signal *f(t)*.This value is described by equation (1).

$$\psi_s = \sqrt{\lim_{T\to 0}\frac{1}{T}\int_{-\frac{T}{2}}^{\frac{T}{2}} f^2(t)dt} \qquad (1)$$

The information related with characteristics of random variables in a time domain is provided by auto-correlation function R_f. The similar information in a frequency domain is given by the power density function S_f.

$$R_f(\tau) = \lim_{T\to 0}\frac{1}{T}\int_{-\frac{T}{2}}^{\frac{T}{2}} f(t)f(t+\tau)dt = \frac{1}{2\pi}\int_{-\infty}^{\infty} S_f(\omega)e^{t\omega\tau}d\omega \qquad (2)$$

(a) First order mode shape

(b) Second order mode shape

(c) Third order mode shape

(d) Fourth order mode shape

(e) Fifth order mode shape

Figure-4. First Five Mode Shapes of Exhaust System

When $\tau = 0$ in time domain;

$$R_f(0) = \lim_{T\to 0}\frac{1}{T}\int_{-\frac{T}{2}}^{\frac{T}{2}} f^2(t)dt = \psi_s^2 = \frac{1}{2\pi}\int_{-\infty}^{\infty} S_f(\omega)d\omega \qquad (3)$$

The root mean square value at the time domain is equivalent to the area under the frequency response curve in the frequency domain [9]. So hanger locations of the exhaust system can be selected by the lowest RMS value.

3.2 Calculating RMS Value

Vibration energy distribution is crucial for determining hanger location of the exhaust system. Based on frequency response function, the different energy distribution at different measuring points of exhaust system can be generated. For an instant, the response at measuring point 40 of the exhaust system is shown in Figure-5 describing acceleration translational magnitude with respect to the frequency. The area under the graph represents the vibrating energy of the given node.

Based on the results, the average excited load response at the different points is identified utilizing RMS value that is the transmitted energy to the body. The RMS value is very simple and easy to observe the average transmitted energy at various measuring points. The highest RMS values give the highest energy.

However, the high energy arise disadvantage for the hanger's locations of the exhaust system because the energy can be transmitted via the hangers to the vehicle body and it may then be arose noise, vibration and harshness problem of the vehicle. Therefore, the small RMS value should be selected for the hanger position. The level of RMS value at the different points is depicted in Figure-6.

3.3 Selecting New Hanger Location

Originally, there are three hanger locations; the first one is at point 10 which is near the flexible bellow, the second is at 1.5 cm away from point 37 which is between measuring point 37and 38, and the third is at 8.5 cm away from measuring point 40 between point 40 and 41 or at the rear part of the muffler. The some original positions are at high vibrating energy that transfers more energy to the body. Therefore, changing the hanger locations is required to reduce the vibration of exhaust system and the body.

Based on the RMS level of vibrating energy, the measuring point 11 should be selected as a first new hanger location because its RMS level is small compare to the surrounding measuring points. The measuring points 16 and 17 have the same RMS value of vibrating energy. These two points are neighbouring points. So, we could not choose both points. On the other hand, the location of measuring point 16 is on the catalytic converter and that of point 17 is on the exhaust pipe near the catalytic converter. Since the exhaust hanger should not be fitted on the catalytic converter, the measuring point 17 was selected as a second new hanger location. The measuring point 23 has lower RMS value of energy but it should not be selected as a hanger location because this point is near to the second new hanger location. The rest point which has lower RMS value of vibrating energy is the measuring point 40. This point is located at the front of the muffler. We select this measuring point as a third new hanger location of the exhaust system.

To sum up, the new selected hanger locations are at measuring point 11, 17 and 40. These are the lowest RMS value which is recommended as the best hanger locations of the sample exhaust system. The original and new hanger locations of the exhaust system are shown in Figure-7.

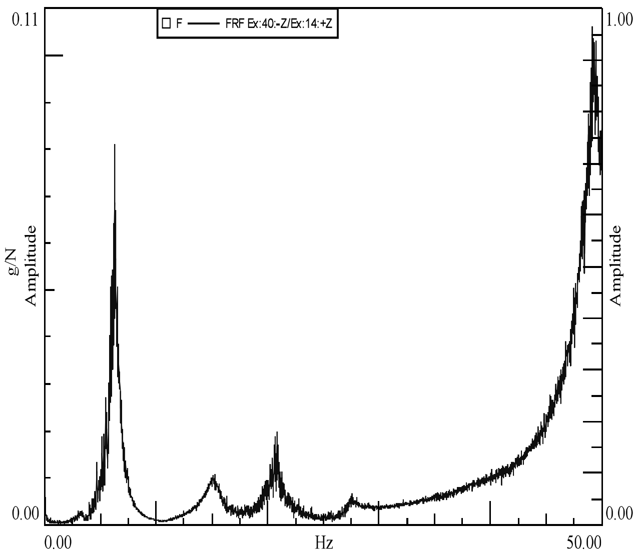

Figure 5. Response of Exhaust System

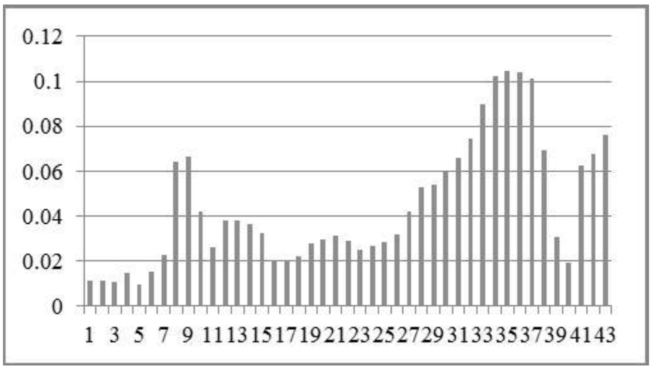

Figure 6. Level of RMS Value at Different Points

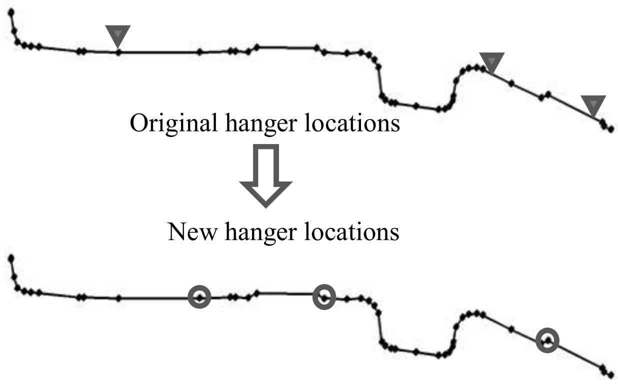

Figure 7. Hanger Locations of Exhaust System

4. CONCLUSION

In this paper, experimental modal analysis was performed to determine the suitable hanger locations of the sample exhaust system using root mean square (RMS) value. The spectral testing software of LMS Test.Lab was used to generate the modal properties of the sample exhaust system. The first five natural frequencies and the relative mode shapes have been generated within the frequency range of interest. Three new hanger locations were selected at the smaller RMS value not only to reduce the vibration of the exhaust system but also to increase durability of the hangers. As a result, the suitable hanger locations of the exhaust system can be investigated easily to reduce the transmitted vibration energy from the exhaust system to the vehicle body that arise the noise, vibration and harshness issue while estimating the dynamic properties of the exhaust system for the development process of the automobile. This technique can be applied as a tool to study and estimate dynamic characteristics of the exhaust system, and to find suitable hanger location of exhaust system easily and quickly.

5. FUTURE WORK

The result of this experimental modal analysis will be validated with the results of numerical simulation by utilizing MSC Patran / Nastran software.

ACKNOWLEDGEMENTS

This study is supported by Japan-ASEAN Integration Fund (JIAF). This financial support is gratefully acknowledged. Then, the author would like to thank to his supervisors for their guidance and encouragement.

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